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Effect of Two Synthetic Lubricants on Life of AISI 9310 Spur Gears

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EFFECT OF TWO SYNTHETIC LUBRICANTS ON LIFE OF AISI 9310 SPUR GEARS

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SUMMARY

Spur-gear surface fatigue tests were conducted with two lubricants using a single lot of consumable-electrode, vacuum-melted (CVM), AISI 9310 spur gears. The gears were case carburized and hardened to Rockwell C60. The gear pitch diameter was 8.89 cm (3.5 in.). The lot of gears was divided into two groups, each of which was tested with a different lubricant. The test lubricants are classified as synthetic polyol-ester-based lubricants. One lubricant is 30 percent more viscous than the other. Both lubricants have similar pressure viscosity coefficients. Test conditions included a bulk gear temperature of 350 K (170 °F), a maximum Hertz stress of 1.71 GPa (248 ksi) at the pitch line, and a speed of 10 000 rpm.

The surface fatigue life of gears tested with one lubricant was approximately 2.4 times that for gears tested with the other lubricant. The lubricant with the 30-percent higher viscosity gave a calculated elastohydrodynamic (EHD) film thickness that was 20 percent higher than the other lubricant. This increased EHD film thickness is the most probable reason for the improvement in surface fatigue life of gears tested with this lubricant over gears tested with the less viscous lubricant.

INTRODUCTION

The surface-pitting fatigue of gears is affected by several variables. The lubricant viscosity, other physical properties, the lubricant extreme-pressure (EP) additives (or boundary film properties), and other chemical properties all influence the surface fatigue life of gears. For aircraft applications and other gearing applications, it is important to know how a given lubricant may affect the surface fatigue life of a gear system. Therefore, when a new or improved lubricant is developed and/or selected for an application, a program to evaluate the fatigue life of the lubricant gear combination should be conducted.

The NASA Lewis Research Center gear test rigs were used to evaluate several lubricants (ref. 1) and lubricant additives (refs. 2 and 3). It was found that lubricants with similar viscosity but with different chemical content gave improved gear fatigue lives that were up to five to one difference. Studies on the effect of additives in a base stock lubricant (ref. 2), found that the lubricant containing a phosphorous-type load-carrying additive improved the gear surface fatigue life by 2.4 times that of the same lubricant with no additive.

In most gear systems the lubricant additives are necessary for the system operation. The additive can minimize or prevent the wear and surface damage resulting from very thin EHD or boundary lubrication conditions. The additives either absorb on the surface film or react with the surface to form a film or coating. This film or coating protects the surface by reducing the friction coefficient and preventing metal to metal contact of the surface asperities (refs. 4 and 5). These results indicate the necessity for conducting gear surface fatigue tests on new lubricant formulations for various applications. These tests results also indicate the effect that various additives in the same viscosity lubricants can have on the surface pitting fatigue life. Other test programs (refs. 6 and 7) have shown the effect of the lubricant viscosity on the surface fatigue life of rolling-element components. The viscosity of the lubricants contributes to the elastohydrodynamic (EHD)-film-forming property, which also changes the surface fatigue life of the component.

The objective of the work reported herein was to determine the effect of two different lubricants on the surface-pitting fatigue life of consumable-electrode, vacuum-melted (CVM), carburized, and hardened AISI 9310 steel spur gears. One lot of spur gears was manufactured from a single heat of CVM AISI 9310 material to accomplish the objective. The gears were all case carburized, hardened, and ground to the same specification. The gear pitch diameter was 8.89 cm (3.5 in.). The lot of gears was divided into two groups, each of which was tested with a different lubricant. The test lubricants were synthetic polyol-ester-based lubricants. The test conditions included a gear bulk temperature of 350 K (170 °F), a maximum Hertz stress of 1.71 GPa (248 ksi) at the pitch line, and a speed of 10 000 rpm.

APPARATUS, SPECIMENS, AND PROCEDURE

Gear Test Apparatus

The gear fatigue tests were performed in the NASA Lewis Research Center's gear test apparatus. The test rig is shown in figure 1(a) and described in reference 8. This test rig uses the four-square principle of applying the test gear load so that the input drive only needs to overcome the frictional losses in the system.

A schematic of the test rig is shown in figure 1(b). Oil pressure and leakage flow are supplied to the load vanes through a shaft seal. As the oil pressure is increased on the load vanes inside the slave gear, torque is applied to the shaft. This torque is transmitted through the test gears back to the slave gear where an equal but opposite torque is maintained by the oil pressure. This torque on the test gear, which depends on the hydraulic pressure applied to the load vanes, loads the gear teeth to the desired stress level. The two identical test gears can be started under no load, and the load can be applied gradually, without changing the running track on the gear teeth.

Separate lubrication systems are provided for the test gears and the main gearbox. The two lubricant systems are separated at the gearbox shafts by pressurized labyrinth seals using nitrogen as the seal gas. The test gear lubricant is filtered through a 5-μm nominal fiberglass filter. The test lubricant can be heated electrically with an immersion heater. The skin temperature of the heater is controlled to prevent overheating the test lubricant.

A vibration transducer mounted on the gearbox is used to automatically shut off the test rig when a gear-surface fatigue spall occurs. The gearbox is also automatically shut off if there is a loss of oil flow to either the main gearbox or the test gears, if the test gear oil overheats, or if there is a loss of seal gas pressurization.

The belt-driven test rig can be operated at several fixed speeds by changing pulleys. The operating speed for the tests reported herein was 10 000 rpm.

Test Materials

The test gears were manufactured from consumable-electrode vacuum-melted (CVM), AISI 9310 steel from the same heat of material. The nominal chemical composition of the material is given in table I. All sets of gears were case carburized and heat treated in accordance with the heat treatment schedule of table II. Figure 2 is a photomicrograph of an etched and polished gear tooth showing the case and core microstructure of the AISI 9310 material. This material had a case hardness of Rockwell C58 and a case depth of 0.97 mm 0.038 in.). The nominal core hardness was Rockwell C40.

Test Gears

Dimensions for the test gears are given in table III. All gears have a nominal surface finish on the tooth face of 0.406 μm (16 μin.) rms and a standard 20° involute profile with tip relief. Tip relief was 0.0013 cm (0.0005 in.), starting at the highest point of single tooth contact.

Test Procedure

After the test gears were cleaned to remove the preservative, they were assembled on the test rig. The test gears were run in an offset condition with a 0.30-cm (0.120-in.) tooth-surface overlap to give a load surface on the gear face of 0.28 cm (0.110 in.), thereby allowing for the edge radius of the gear teeth. If both faces of the gears were tested, four fatigue tests could be run for each pair of gears. All tests were run-in at a load of 1225 N/cm (700 lb/in.) for 1 hr. The load was then increased to 5784 N/cm (3305 lb/in.), which gives a 1.71 GPa (248 ksi) pitch-line maximum Hertz stress. At the pitch-line load, the tooth bending stress was 0.21 GPa (30 000 psi) if plain bending is assumed. However, because of the offset load, an additional stress is imposed on the tooth bending stress. Combining the bending and torsional moments gives a maximum stress of 0.26 GPa (37 000 psi). This bending stress does not include the effects of tip relief which would also increase the bending stress. In general, 20 tests were run for each lubricant.

Operating the test gears at 10 000 rpm gave a pitch-line velocity of 46.55 m/sec (9163 ft/min). Lubricant was supplied to the inlet mesh at

800 cm^3/min at 320±6 K (116±10 °F). The lubricant outlet temperature was nearly constant at 350±3 K (170±5 °F). The tests ran continuously (24 hr/day) until they were automatically shut down by the vibration detection transducer located on the gearbox adjacent to the test gears. The lubricant circulated through a 5-μm fiberglass filter to remove wear particles. After each test the lubricant and filter element were discarded. Inlet and outlet oil temperatures were continuously recorded on a strip-chart recorder. After each test the system was partially disassembled, flushed with trichloroethane and then with alcohol, dried, and reassembled before a new lubricant was used.

Test Lubricant

Two lubricants were used for endurance tests with the AISI 9310 gear test specimens. Both are classified as polyol-ester synthetic lubricants. A summary of the properties of these lubricants is given in table IV. The Navy Air Propulsion Center (NAPC) codes for the lubricant are given in the table. The additive and chemical data for these lubricants are proprietary to the respective manufacturers. It is expected that both lubricants would have antiwear additives as well as oxidation inhibitors.

The reference lubricant PE-5-L-1274 is used in the program to compare with other lubricants. The lubricants PE-5-L-1307 and PE-5-L-1553 are the same lubricant, but two different batches were used. Both of these last two lubricants meet the military MIL-L-23699 lubricant specification.

The pitch-line elastohydrodynamic (EHD) film thickness was calculated for each of the lubricants by using the method described in references 9 and 10 and the data of table IV. It was assumed for this film-thickness calculation that the gear temperature at the pitch line was equal to the oil outlet temperature and that the oil on the surface of the gear tooth entering the contact zone was equal to the gear temperature, even though the oil inlet or oil jet temperature was considerably lower. The gear surface temperature may be somewhat higher than the oil outlet temperature, especially at the end points of sliding contact. The computed EHD film thicknesses are given in table IV as are the initial Λ ratios (film thickness divided by composite surface roughness (h/σ) at the 1.71-GPa (248-ksi) pitch-line maximum Hertz stress). Based on the Λ values, the lives of the gears with each lubricant would not be expected to be significantly different except when additives became important (ref. 3).

RESULTS AND DISCUSSION

The gear fatigue life for each of the two lubricants is shown in figure 3 and is summarized in table V. The surface-pitting fatigue life of the AISI 9310 gears run with lubricant A, a polyol-ester synthetic oil, is shown in figure 3(a). These data, which are shown on Weibull coordinates, were analyzed by the method described in reference 11. The life shown is that of gear pairs failed in millions of stress cycles. The gear teeth receive one stress cycle per revolution. A failure is defined as one or more spalls covering more than 50 percent of the width of the tooth Hertzian contact. A typical fatigue spall is shown in figure 4(a); a cross section of the spall is shown in figure 4(b). The failure index in figure 3 indicates the number of failures out of the number of tests run. For lubricant A, the failure index was 30 out of 30. The

10- and 50-percent system lives (these are the lives at a 90- and 50-percent probability of survival, respectively) were 5.1 and 20.4 million revolutions or stress cycles, respectively.

The surface-pitting fatigue life of the CVM AISI 9310 steel spur gears run with lubricant B, also a polyol-ester synthetic oil, is shown in figure 3(b). A typical fatigue spall for lubricant B is shown in figure 5(a); a cross section of the spall is shown in figure 5(b). The 10- and 50-percent system lives with this fluid were 12.1 and 76 million stress cycles, respectively. The 10-percent life of lubricant B is nearly 2.4 times greater than the life obtained for lubricant A. The differences in these lives, based on the confidence numbers given in table V, are statistically significant. (The confidence number indicates the percentage of time the order of the test results would be the same. For a confidence number of 84 percent, lubricant B will produce a higher life than lubricant A 84 out of 100 tests. Generally, a 2-σ or a 95-percent confidence is considered statistically significant. However, experience has shown that a confidence number of 80 percent or greater is necessary to draw useful conclusions regarding life differences.)

The difference in surface fatigue life between lubricants A and B can be partially explained by the difference in viscosity between the two lubricants. The viscosity of lubricant B is 30 percent higher than that of lubricant A. This higher viscosity gives a minimum EHD film thickness that is 20 percent higher for lubricant B, assuming that both lubricants have the same pressure viscosity coefficient. Since both lubricants have similar base stocks, one would expect them to have nearly identical pressure viscosity coefficients. It is shown in reference 12 that the EHD lubricant film thickness can affect the depth to the maximum subsurface shear stress. Tests with bearings using lubricants that produce different film thicknesses have shown improved surface fatigue life with increased EHD film thickness (ref. 13). The EHD film thickness for these tests was in the range where small changes in EHD film thickness can cause significant changes in the surface fatigue life (ref. 14).

Another factor contributing to differences in surface fatigue life in gears is the type of additive used in the lubricant. In reference 2 it was demonstrated that a phorphorous-type additive can improve the surface fatigue life of gears. Because the type and amount of additives in these lubricants is unknown, it cannot be determined from this test whether the additives in these lubricants effected the gear life.

SUMMARY OF RESULTS

Spur-gear surface-pitting fatigue tests were conducted with two lubricants using a single lot of consumable-electrode, vacuum melted (CVM), AISI 9310 spur gears. The gears were case carburized and hardened to Rockwell C60. The gear pitch diameter was 8.89 cm (3.5 in.). The lot of gears was divided into two groups, each of which was tested with a different test lubricant. The test lubricants are classified as synthetic polyol-ester-based lubricants. Although lubricant B is 30 percent more viscous than lubricant A, both have similar pressure viscosity coefficients. Test conditions included a bulk gear temperature of 350 K (170 °F), a maximum Hertz stress of 1.71 GPa (248 ksi) at the pitch line, and a speed of 10 000 rpm. The following results were obtained:

1. The surface fatigue life of gears tested with lubricant B was approximately 2.4 times the life of gears tested with lubricant A.

2. The 30-percent higher viscosity of lubricant B over that of lubricant A gave a calculated elastohydrodynamic (EHD) film thickness for lubricant B that is 20 percent higher than for lubricant A. This increased film thickness is the most probable cause for the improvement in surface fatigue life of gears tested with lubricant B over that for gears tested with lubricant A.

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TABLE I. - NOMINAL CHEMICAL COMPOSTION OF AISI 9310 GEAR MATERIAL

[Composition, wt %.]

Chemical	C	Mn	Si	Ni	Cr	Mo	Cu	P	S
wt %	0.10	0.63	0.27	3.22	1.21	0.12	0.13	0.005	0.005

TABLE II. - HEAT TREATMENT FOR AISI 9310

Step	Process	Temperature		Time, hr
		K	°F	
1	Preheat in air	----	----	------
2	Carburize	1172	1650	8
3	Air cool to room temperature	----	----	------
4	Copper plate all over	----	----	------
5	Reheat	922	1200	2.5
6	Air cool to room temperature	----	----	------
7	Austenitize	1117	1550	2.5
8	Oil quench	----	----	------
9	Subzero cool	180	-120	3.5
10	Double temper	450	350	2 each
11	Finish grind	----	----	------
12	Stress relieve	450	350	2

TABLE III. - SPUR-GEAR DATA

[Gear tolerance per ASMA class 12.]

Number of teeth	28
Diametral pitch	8
Circular pitch, cm (in.)	0.9975 (0.3927)
Whole depth, cm (in.)	0.762 (0.300)
Addendum, cm (in.)	0.318 (0.125)
Chordal tooth thickness reference, cm (in.)	0.485 (0.191)
Pressure angle, deg	20
Pitch diameter, cm (in.)	8.890 (3.500)
Outside diameter, cm (in.)	9.525 (3.750)
Root fillet, cm (in.)	0.102 to 0.152 (0.04 to 0.06)
Measurement over pins, cm (in.)	9.603 to 9.630 (3.7807 to 3.7915)
Pin diameter, cm (in.)	8.549 (0.216)
Backlash reference, cm (in.)	0.0254 (0.010)
Tip relief, cm (in.)	0.001 to 0.0015 (0.0004 to 0.0006)
Surface finish, μm (μin.)	0.406 (16)

TABLE IV. - LUBRICANT PROPERTIES

Lubricant			
NASA identification	A	B	
NAPC code number	PE-5-L-1274	PE-5-L-1307	PE-5-L-1553
Kinematic viscosity at -			
311 K (100 °F)	21.0	30.27	29.11
373 K (210 °F)	4.31	5.46	5.32
Flash point K (°F)	516 (470)	539 (510)	539 (510)
Pour point K (°F)	<200 (<-100)	220 (-64)	213 (-76)
Specific gravity at -			
298 K (77 °F)	0.998	-----	
289 K (60 °F)	-----	1.000	
Total acid number, tan mg KoH/g oil	0.07	0.03	
Elastohydrodynamic film thickness, h, μm (μin.)	0.432 (17)	0.518 (20.4)	
Λ ratio (h/σ)	0.75	0.90	

TABLE V. - SURFACE-PITTING (ROLLING-ELEMENT) FATIGUE LIVES

[NASA spur-gear test apparatus; material, CVM AISI 9310; gear bulk temperature, 350 K (170 °F); maximum hertz stress, 1 GPa (248 ksi); speed, 10 000 rpm.]

Lubricant code		Lubricant basestock	Gear system life, millions of stress cycles		Weibull slope	Failure index[a]	Confidence number,[b] percent
NASA	NAPC		10 percent	50 percent			
A	PE-5-L-1274	Polyol-ester	5.1	20.4	1.36	30 out of 30	--
B	PE-5-L-1307 PE-5-L-1553	Polyol-ester	12.1	76	1.02	20 out of 20	84

[a]Number of failures out of number of tests.

[b]Percent of time that 10-percent life obtained with each lubricant will have the same relation to the 10-percent life of lubricant NASA F.

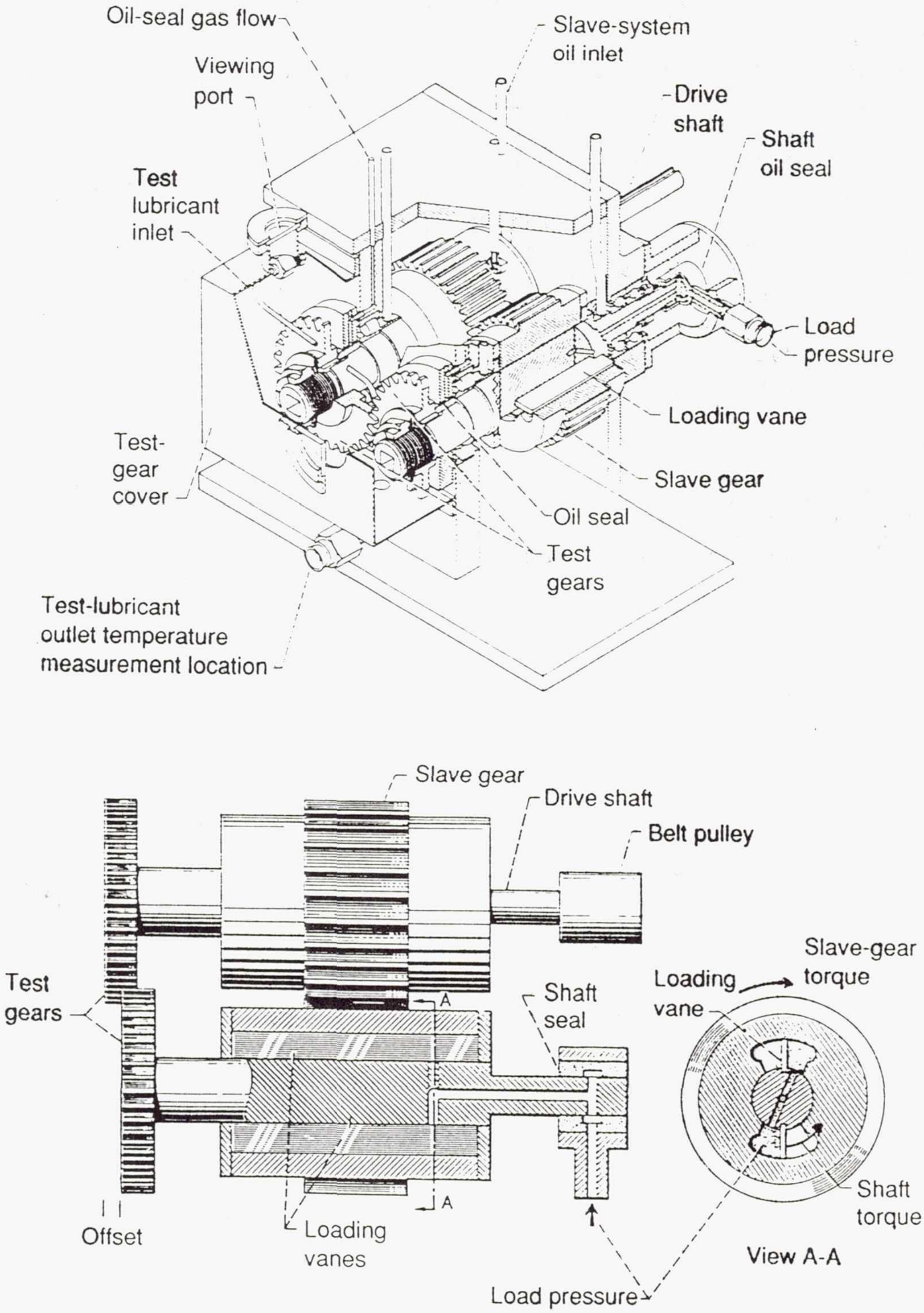
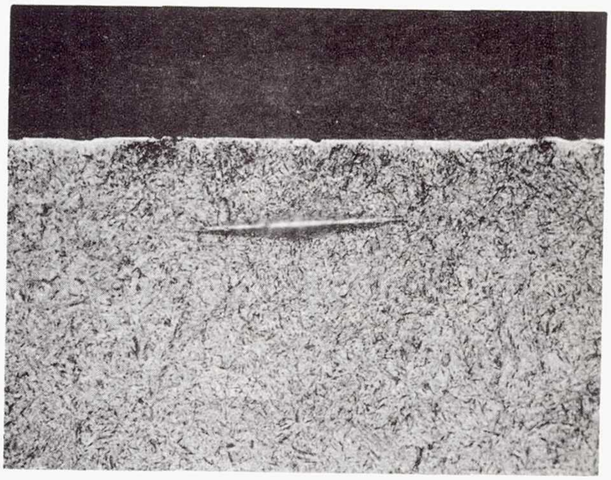
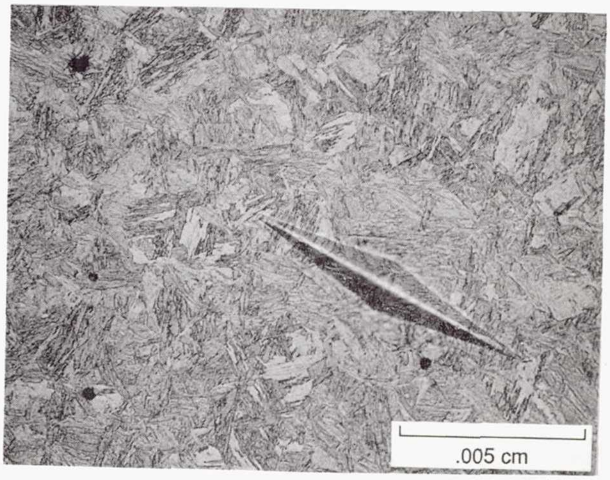


Figure 1.– NASA Lewis Research Center's gear fatigue test apparatus.



(a) Case.



(b) Core.

Figure 2.— Photomicrographs of case and core of CVM AISI 9310 spur gears.

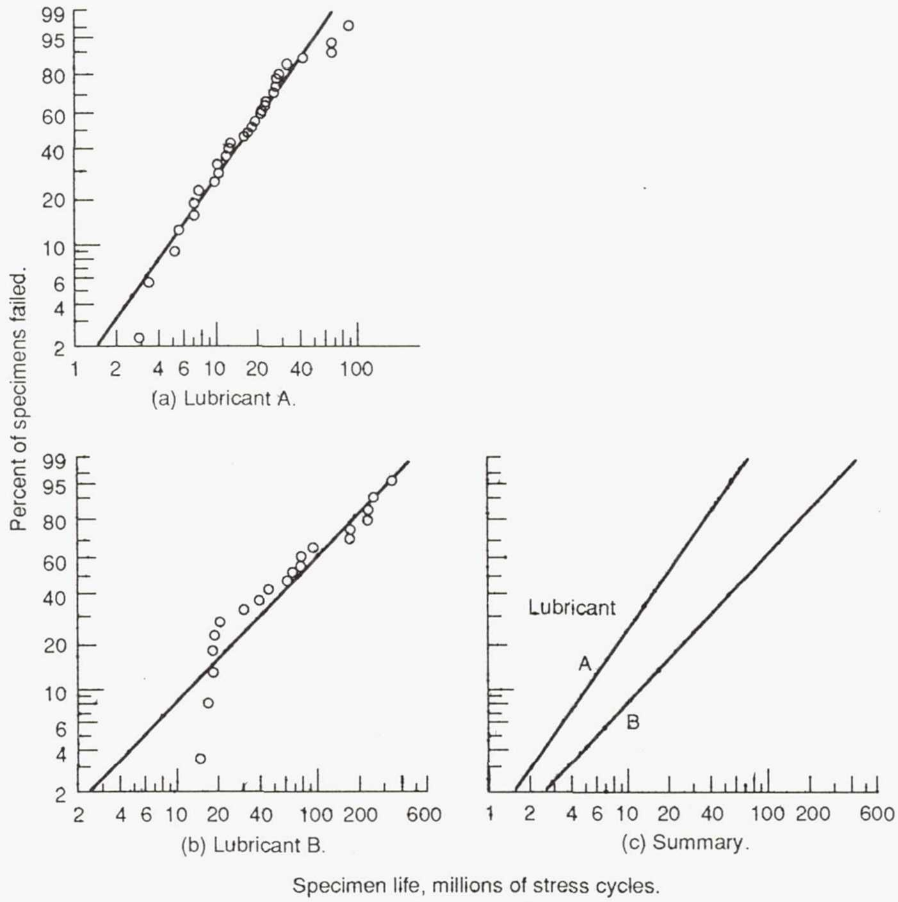
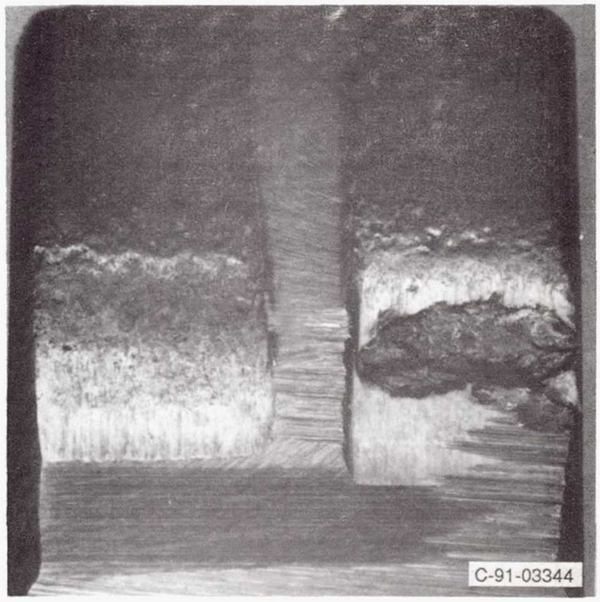
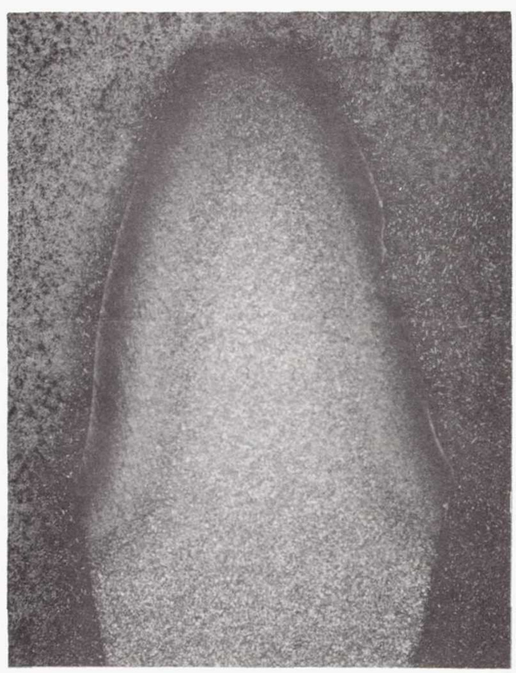


Figure 3.—Surface-pitting fatigue lives of CVM AISI 9310 spur gears run with two different lubricants. Gear pitch diameter, 8.59 cm (3.5 in.); speed, 10 000 rpm; maximum hertz stress, 1.71 GPa (248 ksi); gear temperature, 350 K (170 °F).

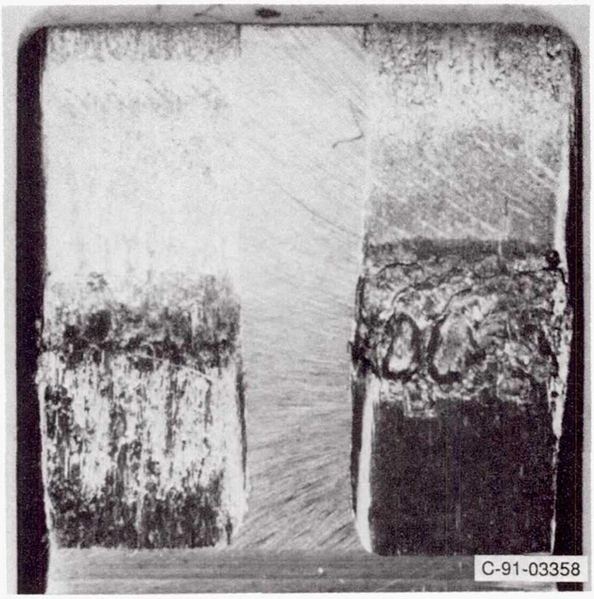


(a) Typical fatigue spall.

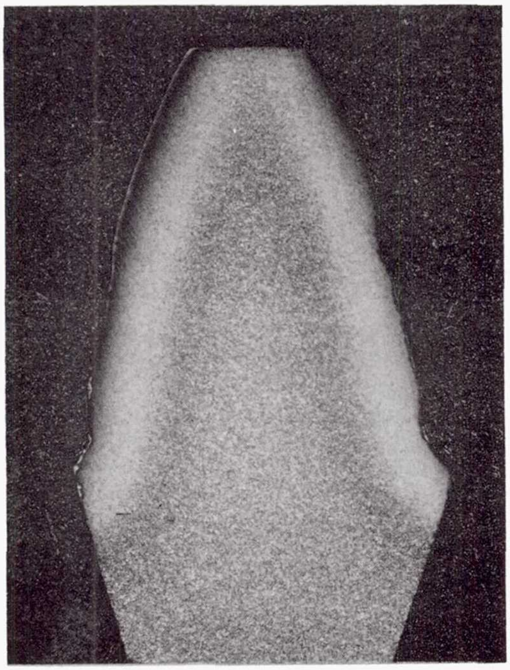


(b) Cross section of typical fatigue spall.

Figure 4.— Fatigue spall for lubricant A.



(a) Typical fatigue spall.



(b) Cross section of typical fatigue spall.

Figure 5.— Fatigue spall for lubricant B.

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15. Supplementary Notes

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16. Abstract

Spur-gear surface fatigue tests were conducted with two lubricants using a single lot of consumable-electrode vacuum-melted (CVM) AISI 9310 spur gears. The gears were case carburized and hardened to Rockwell C60. The gear pitch diameter was 8.89 cm (3.5 in). The lot of gears was divided into two groups, each of which was tested with a different lubricant. The test lubricants can be classified as synthetic polyol-ester-based lubricants. One lubricant was 30 percent more viscous than the other. Both lubricants have similar pressure viscosity coefficients. Test conditions included a bulk gear temperature of 350 K (170 °F), a maximum Hertz stress of 1.71 GPa (248 ksi) at the pitch line, and a speed of 10 000 rpm. The surface fatigue life of gears tested with one lubricant was approximately 2.4 times that for gears tested with the other lubricant. The lubricant with the 30-percent higher viscosity gave a calculated elastohydrodynamic (EHD) film thickness that was 20 percent high than the other lubricant. This increased EHD film thickness is the most probable reason for the improvement in surface fatigue life of gears tested with this lubricant over gears tested with the less viscous lubricant.

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